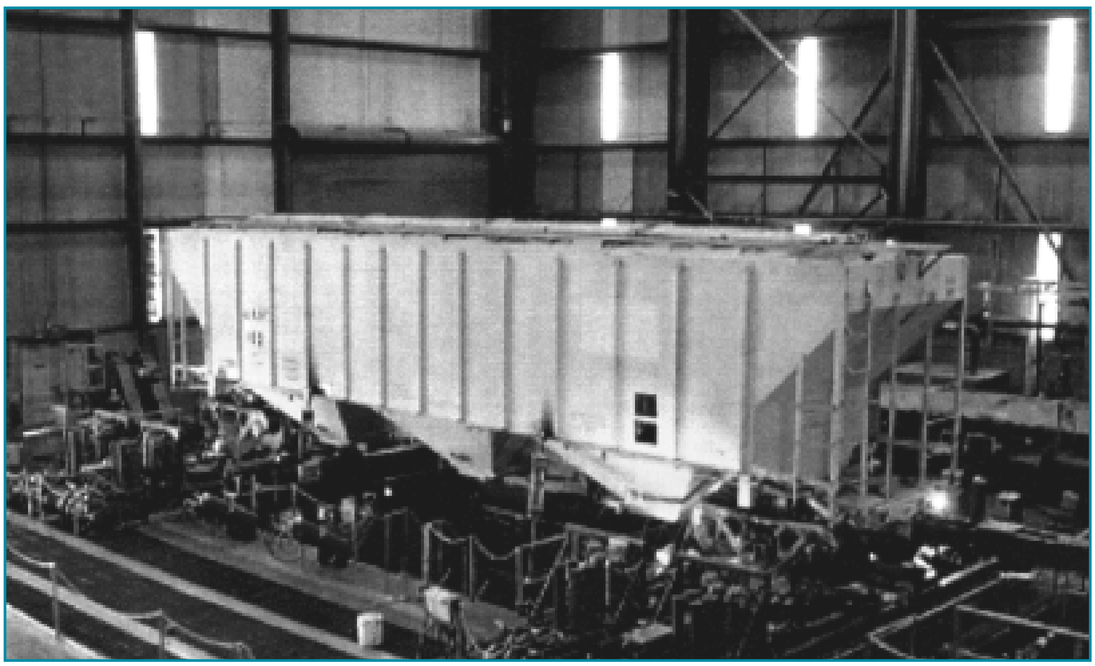
VEHICLE/TRACK INTERACTION



VEHICLE/TRACK INTERACTION

The principal objective of the Vehicle/Track Interaction research effort is to quantify rail vehicle response to track geometry in order to **develop improved approaches to track geometry inspection and maintenance** that are both cost effective and safety enhancing, to **develop modifications to vehicles susceptible to derailment**, and to **develop methodologies for evaluating new vehicle designs** for safe dynamic behavior.

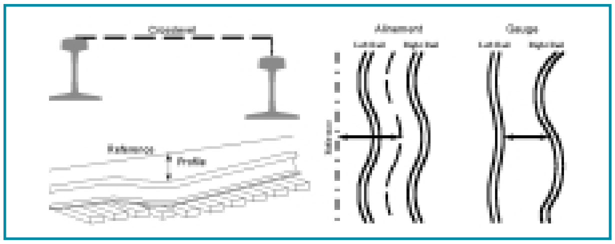
The program to conduct studies of vehicle response to track geometry was developed by the FRA. In addition, some of the results of these efforts are used in studies of gage restraint, track buckling, and rail fatigue. This program is being conducted through cooperative research efforts with the industry and has promoted information exchange among members of the industry and the Government.

Benefits from this FRA Vehicle/Track Interaction Program, covering the major track failure modes, can be expected on several levels. The most important will be fewer unexplained derailments caused by adverse interaction of track, vehicles and operations. Additionally, the research results will provide information necessary to improve industry specifications and recommended practices, as well as aid the FRA in possible rulemaking activities. Further, results have already been incorporated into several railroads' operating and maintenance practices promoting safer track, equipment, and operations.

Track surface geometry is described by **track profile** and **crosslevel**. **Rail profile** is the elevation of the rail relative to a fixed reference line. **Track profile** is the average of the left and right rail profiles while **track crosslevel** is the difference between the left and right rail profiles. **Track alignment** — the direction or 'route' of the track — and **gage** — the distance between the two rails — are required to completely describe track geometry. Track surface and alignment characteristics vary with distance along the track. Because of the nature of track construction, track geometry variations can be repetitive or can be isolated single events.

TRACK GEOMETRY - VARIATIONS

Excessive variations in any of the four track geometry characteristics can lead to a derailment.



In addition, track geometry variations can cause large lateral rolling and vertical bounce motions of vehicles, and can induce large lateral and vertical forces between the wheel and the rail. These motions and forces can be oscillatory. The motions and forces might vary as the vehicle travels on the track, causing buildup of resonant motions of the vehicle, or they might be single events, occurring only once at a particular track location as each vehicle passes.

RESEARCH STATUS

The initial approach used in the program to study track geometry emphasized use of derailment scenarios that could be associated with a high number of accidents caused by track geometry variations and irregularities. As a result, these studies focused on harmonic roll associated with high center of gravity cars operating on half-staggered bolt-jointed rail of 39-foot lengths. This scenario is characterized by a low-speed (10 to 20 mph) derailment of a car having a truck center spacing of less than 45 feet. Both government and industry developed simulation programs to predict harmonic roll response. Predictions made with these simulation programs have shown good agreement with field and laboratory test data.

After developing an understanding of harmonic roll, the approach was broadened to include all scenarios involving track surface geometry. The conditions studied include car body rollover due to the harmonic roll response of freight vehicles to repeated crosslevel variations, wheel climb derailment and carbody/truck separation due to track twist, wheel climb derailment due to excess superelevation in curves, and carbody/truck separation and truck loading in excess of design limits due to the harmonic bounce response of a rail vehicle to repeated track profile variations.

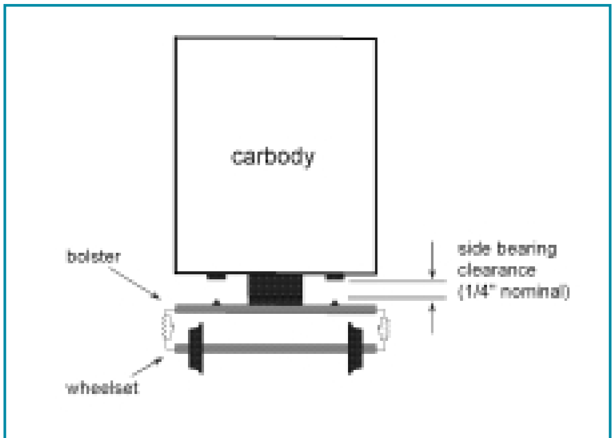
The FRA has developed algorithms based on the response of an idealized vehicle model to the track surface geometry. These algorithms can distinguish between a single geometry perturbation with a relatively large amplitude, which is a safe track condition, and repeated geometry perturbations each with a relatively small amplitude, which is an unsafe track condition. The algorithms require multiple measurements and extensive computations before comparison to maximum values. The algorithms can identify track segments having small amplitude periodic irregularities capable of producing resonant response that could produce wheel lift, centerplate separation or coupler separation, without rejecting track that is consistent with current good practice and has been demonstrated to be safe.

The track surface geometry algorithms have been implemented in a computer program and applied to typical track geometry car data. This program has been used to analyze data taken with the T-10 inspection car, operated by the FRA, as well as with railroad track geometry cars. Currently, the algorithms are being implemented in a system which can measure and evaluate track surface geometry in real-time. The real-time implementation of the track surface geometry measurement and evaluation system is being accomplished by updating and refurbishing the T-6 track geometry measurement instrumentation, acquiring the necessary computer hardware and software for evaluation, storage, and display of the track geometry data, and development of the software necessary for data acquisition and display. The track surface geometry algorithms require track alignment, crosslevel, and profile for evaluation.

A typical freight car suspension is provided by 3-piece freight trucks at either end of the carbody, comprised of a bolster, 2 side frames, and 2 axles. The carbody rests on the bolster at the centerplate, and is able to pivot about the edges of the centerplate. Side bearings on either side of the centerplate provide a stop to relative roll between the carbody and the bolster. Side-bearing clearance is the clearance between the carbody and the bolster at the side bearing. Each end of the bolster rests in a side frame, supported vertically by the spring group, in parallel with some auxiliary friction device, usually a snubber.

KEY FINDINGS

In studies conducted by the Office of Railroad Development, it was found that some light weight freight cars traveling over track with large twist (difference in crosslevel between truck centers) in curves would experience lateral-to-vertical force ratios on the lead outer wheels of the truck that were in excess of established wheel climb derailment criteria. These results were confirmed by AAR tests on the "bunched spiral" associated with the tests for New and Untried Cars prescribed by Chapter XI of the AAR Manual of Standards and Recommended Practices. An option for increasing the tolerance of freight cars to track twist is to increase the side-bearing clearance. However, increasing the side-bearing clearance has the potential for changing the roll response of a freight car to repeated crosslevel variations. Initial simulation studies indicated that increasing side-bearing clearance could also have the beneficial effect of reducing wheel unloading and reducing carbody roll response. This result was contrary to railroad industry experience and earlier studies. In order to resolve the differences between the initial numerical simulation results and the industry experience and to provide an improved calibration of a simulation model for predicting the influence of side-bearing clearance on freight car roll response, a series of tests were performed at the TTC in Pueblo, Colorado. Tests on a loaded 100-ton hopper car were conducted on the Vibration Test Unit (VTU) and the Precision Test Track (PTT) at TTC in September and October 1993.



Dynamics of Wheel Climb

In 1994, the AAR and FRA began a jointly-funded research program to examine the mechanics of wheel climb (also called flange climb) derailments. The AAR conducted the tests using its Track Loading Vehicle (TLV) at the TTC. The primary objective of this testing was to reexamine the current wheel climb criteria used in Chapter XI of the AAR's Manual of Standards and Recommended Practices and was the first time full-scale testing of wheel climb had been performed in North America.

During testing, controlled wheel climb derailments of an instrumented test wheel set were achieved under a range of applied wheel/rail forces,wheel set angles of attack, rail profiles, and lubrication conditions.

TLV Wheel Climb Test

RESEARCH STATUS

From this testing, the following conclusions have been drawn:

- No changes are proposed to the existing Chapter XI wheel climb derailment limits.

- The wheel/rail coefficient of friction, the maximum wheel/rail contact angle, and the wheel set angle of attack have a major influence on the potential for wheel climb. Small (or negative) axle angles of attack tend to inhibit wheel climb behavior.
- Geometry at the wheel/rail interface is related to the required flanging wheel lateral/vertical force ratio (L/V) only through the maximum wheel/rail contact angle. In this study, the peak contact angle was similar between an AAR-1B wheel profile on both new and curve-worn rails. Therefore, the L/V needed for wheel climb was equal on both new and worn rail test zones.
- Unlike the L/V ratio, the distance required for a wheel to climb is related to the entire wheel/rail geometry. Interaction for a new geometry typically involves large contact angles being active over a significant amount of lateral wheel shift. A worn wheel/rail contact situation may involve large contact angles for only a small amount of lateral wheel shift. In these worn cases, a large L/V event of significantly shorter duration may lead to wheel climb.
- At zero axle angles of attack, currently used criteria have at least 15% conservatism built into them. At angles of attack greater than +15 mrad (0.8º) the current criteria accurately predict test results.
- All tests and New and Untried Cars Analytical Regime Simulations (NUCARS) converged to the current derailment criteria at higher angles of attack (10-15 mrad). High flanging-rail friction during test Series 10 resulted in axle L/V ratios at wheel climb that were lower than the Chapter XI limit of 1.5.
- The flanging wheel L/V ratio necessary to produce a wheel climb is independent of the friction on the non-flanging rail. Although friction on the non-flanging rail may help create a lateral force on the flanging wheel, this non-flanging friction will not affect the critical L/V which must be achieved before a wheel will climb.
- Vertical load unbalance does not affect the critical L/V values. Again, such an imbalance may create lateral forces on the flanging wheel, but the imbalance will not affect the critical L/V which must be achieved before a wheel will climb.

KEY FINDINGS

It is difficult to control or maintain a constant friction coefficient for any given series of TLV tests. Friction varied from day to day. Future test series to examine critical comparisons of analytical and experimental L/V ratios should be conducted with a constant value of coefficient of friction. Care in cleaning and/or sanding of the rails is very important. Furthermore, after such cleaning, a few break-in derailments should probably be run on the rail surface before beginning a test series. If time permits, additional repetitions of high angle of attack runs should be interspersed within the series to statistically improve the estimate of friction.

FUTURE RESEARCH

Testing and analysis by the AAR are ongoing at the TTC using the TLV. This effort has been designed to improve the current level of understanding of wheel climb behavior. Technical papers were published and presentations were made on the work at the spring 1997 Joint Railroad Conference of the Institute of Electrical and Electronic Engineers and the American Society of Mechanical Engineers.

Vehicle Dynamics Training Module

The results of research related to the mechanics of rail vehicle derailments are being applied to development of educational material to provide railroad and FRA personnel with an improved understanding of requirements for track and equipment maintenance to ensure safe operations.

RESEARCH STATUS

The AAR/TTCI, under direction of the FRA, has produced an educational/safety training video which explains, in layman's language, the mechanics of three categories of freight car derailments: catastrophic, vehicle/track interaction, and human factors. The audiences targeted by this video include railroad operating personnel, track crews, and FRA inspectors. In particular, this video focuses on:

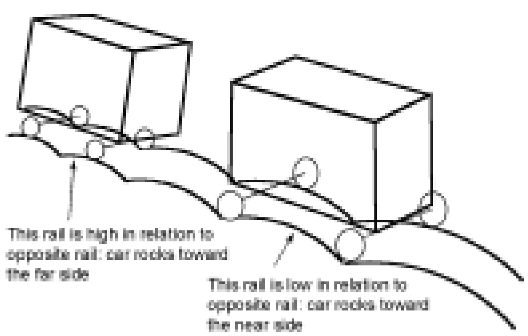
- Basic Dynamics
- Wheel/Rail Interaction
- Wheel/Rail Profiles
- Lubrication Effects
- Car and Truck Design Issues
- Track Geometry Effects
- Track Strength Considerations
- Train Dynamics

This video includes footage of actual derailments and animation generated by the AAR's New and Untried Cars Analytical Regime Simulation (NUCARS) and Train Operation and Energy Simulation (TOES) modeling software. This training video is intended for use as an educational tool in the investigation of derailments, particularly those where no clear causes are evident.

CAR BODY ROLL RESPONSE

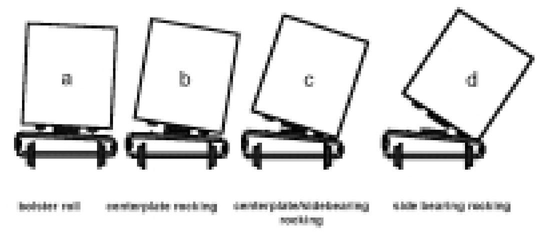
Roll is the rotation of the car body about a longitudinal axis in response to crosslevel variations.

The figure schematically illustrates a car on 39-foot bolt-jointed track with half-staggered low joints. The car starts out with the wheels on one side elevated in relation to the wheels on the opposite side. As the car moves forward, the wheels on one side are lowered while the wheels on the opposite side are raised. At some speeds, the suspension will act to amplify this rolling motion of the carbody, and if the amplitude of the crosslevel variation is sufficient, wheel lift and carbody rollover can occur.



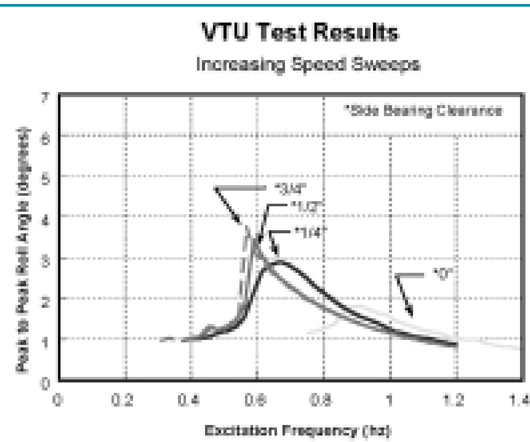
The harmonic roll problem is governed by the non-linear characteristics of the system. Non-linearities arise due to the various support configurations that exist as the carbody extends through its entire range of roll. As the vehicle undergoes harmonic roll, four main roll configurations exist with different roll-moment characteristics corresponding to each roll configuration, resulting in a non-linear effective roll stiffness.

The figure shows the four different roll configurations of a typical freight car. The first configuration, **bolster roll**, occurs when the carbody and bolster rotate together, with no rotation about the edges of the centerplate. **Centerplate rocking** occurs when the carbody rocks about the edges of the centerplate, the extent of this region being dependent upon side-bearing clearance. Further rotation results in **centerplate/side-bearing** rocking as the carbody contacts the side bearings, further displacing the spring groups. **Side-bearing rocking** occurs when the centerplate completely separates from the bolster, with the carbody rotating about the side-bearing.

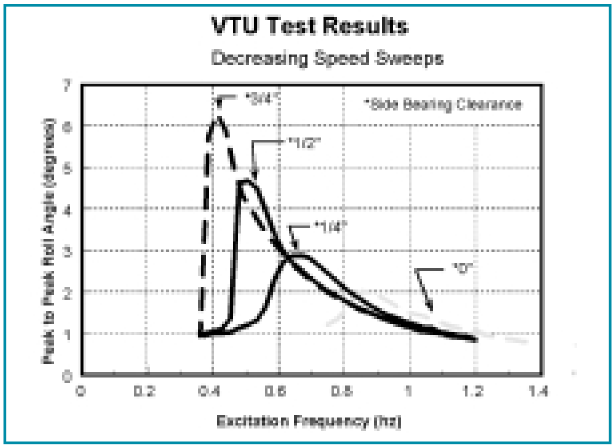


RESEARCH STATUS

The figure shows a comparison plot of roll response for **increasing sweep** (increasing speed) at side-bearing clearances of 0, 1/4, 1/2, and 3/4 inches with input amplitude of 3/8-inch. The plot shows that the limiting response for increasing speed has been reached at 1/2-inch side-bearing clearance, and further increasing side-bearing clearance to 3/4-inch exhibits the same response. For increasing speed, this figure indicates that side-bearing clearance beyond 1/4-inch has only a small influence on maximum carbody roll angle.



The figure shows a comparison plot of roll response for **decreasing sweep** (decreasing speed) at side-



bearing clearances of 0, 1/4, 1/2, and 3/4 inches with input amplitude of 3/8-inch. This plot shows how the car's behavior differs dramatically as speed decreases, from its behavior with speed increasing. Evident is the increased roll angle as side-bearing clearance is increased, because the softening effect associated with centerplate rocking extends the centerplate rocking region to lower frequencies. Carbody response will continue to increase with decreasing speed in this region, until response jumps down to the linear bolster roll region. No limiting response is reached, with the response being substantially greater at 3/4-inch side-bearing clearance than the response at 1/2-inch side-bearing clearance. For decreasing speed, this figure indicates that side-bearing clearance beyond 1/4-inch has a strong influence on maximum carbody roll angle.

KEY FINDINGS

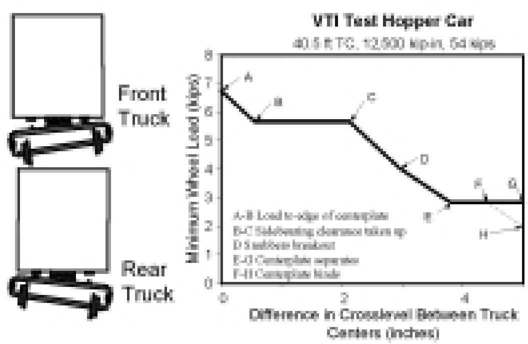
Results of the vibration tests clearly demonstrated the "jump" phenomenon associated with the non-linear behavior of freight car response to crosslevel variations. Tests on the VTU simulating the freight car traversing a series of repeated crosslevel variations while *decreasing* speed produced a much greater roll response than experienced in traversing the same perturbations at *increasing* speed. Tests on the VTU at decreasing speed showed that as side-bearing clearance is increased, carbody roll angle increases.

On-track tests were limited to constant speed runs due to the length of the test section. On-track results show that both 1/4-inch and 3/4-inch side-bearing clearance configurations exhibited wheel lift and maximum peak to peak roll angle in excess of 6 degrees. In addition, the 3/4-inch side-bearing configuration appeared to be sensitive to track anomalies, as evidenced by the sudden jump in response at low speeds. Computer simulations, using a revised model, representing traversal over repeated 3/4-inch low joints confirmed that response at decreasing speed is worse than response at increasing speed. Additionally, as a result of this testing program, it was found that both VTU and on-track testing methods have advantages and disadvantages. Computer simulations can help resolve many issues and provide insight to certain phenomena, but they should not be used alone.

TRACK TWIST

Curved track is usually superelevated, with the outside rail on a curve higher than the inside rail. In order to obtain the superelevation required for the curve, the outside rail is gradually raised, thereby "warping" or "twisting" the track.

Although "twist" is a design feature, irregularities in the track can cause the twist at the entry and exit of a curve to be greater than designed. Track twist can also occur unintentionally, caused by defects in the track. In addition to causing a crosslevel irregularity, a single low joint causes the track to be twisted.



When a vehicle in good condition is on level track, all the wheels equally share the load. When the same vehicle is on twisted track, the wheel loads are redistributed. The situation is somewhat analogous to a table on a warped floor, which has a tendency to rock between diagonally opposite legs. The suspension of rail vehicles allows them to negotiate some amount of track twist without excessive changes in the load supported by the wheels. The figure above shows model predictions of the change in vertical load for a hopper car due to track twist.

In addition to carrying the weight of the vehicle, the wheels must also transmit the lateral loads required for the vehicle to negotiate the curve. These lateral loads for curve negotiation can be quite high, even at low vehicle speeds. A wheel with insufficient vertical load and a high lateral load can be forced up and over the top of the rail, thus derailing.

RESEARCH STATUS

Analytical models have been developed to study rail vehicle response to track twist. Tests have recently been completed at the TTC to ensure that the analytical model can truly predict the behavior of the vehicle and to experimentally determine the safe limits of track twist. The track situations tested included Chapter XI 'Bunched Spiral' and the 'Limiting Spiral,' entry and exit spirals, curved track with twist perturbation, curved track with alignment perturbation and twist perturbation, tangent track with twist perturbation, and tangent track with alignment perturbation and twist perturbation.

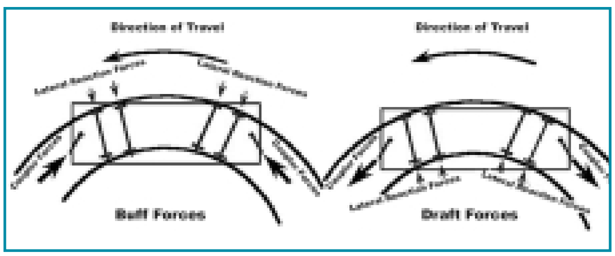
The approach was to test three vehicles which are predicted to be susceptible to wheel unloading due to track twist on tangent, spiral, and curved track with twist perturbations. Wheel unloading as a function of the difference in crosslevel between truck centers was measured during jacking tests performed before the on-track testing. These jacking test measurements were used to verify and adjust the track conditions to be tested. After the first series of tests had been run, a second series was run in which an alignment perturbation was added to the track in addition to the twist perturbation.

LATERAL AND VERTICAL TRAIN FORCES

Train handling produces longitudinal train forces owing to train action as the train is accelerated and braked.

These train forces can result in significant lateral forces when the train traverses a curve. During braking, buff forces can increase the lateral forces acting on the high rail sufficiently to allow a wheel to climb the high rail, and during acceleration, draft forces can increase the lateral forces acting on the low rail sufficiently to cause the low rail to roll over.

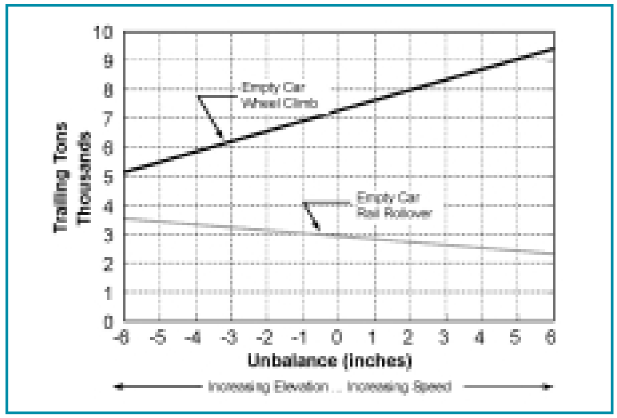
The figure shows buff and draft forces and their lateral reaction forces. The longitudinal train forces, and hence the lateral forces, are dependent on the trailing tonnage.



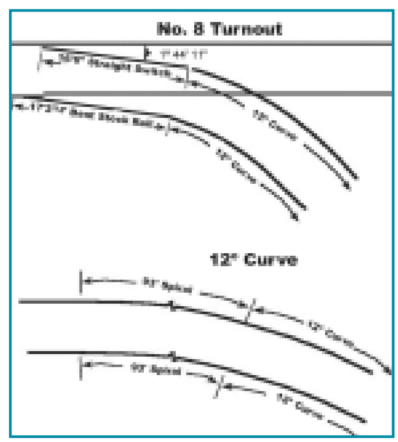
The lateral force that results from the longitudinal train force acts to redistribute the vertical wheel loads and generates a lateral load on the track. In addition to the train forces, the vertical forces are also influenced by the superelevation of the curve and the train speed. This shift in vertical forces comes about principally due to the inertial forces acting on the center of gravity of the car and to the deflection of the suspension springs.

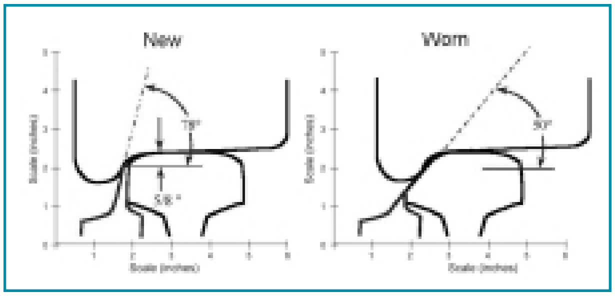
RESEARCH STATUS

Maximum trailing tonnage was determined as a function of elevation for the unloaded car traversing a 6-degree curve on a 2 percent grade. Results are shown in the figure for two cases, wheel climb and rail rollover for the empty car. The empty car is the most critical car in both cases. Since both wheel climb and rail rollover become likely to occur at critical L/V ratios, the lower the magnitude of the vertical force, the lower the magnitude of the lateral force required to cause the critical condition.



Derailment can occur when a car traverses a switch if there is excessive wear of the gage face of the switch rail. Lateral wheel/rail forces are developed as the car traverses the curvature of the turnout, allowing a wheel to climb the rail if the contact angle between the wheel and the rail is sufficiently shallow. The figure shows a sketch of the track route alignment geometry for a No. 8 turnout. The low rail in the curve transitions directly from tangent track to curved track. The diverging point of the switch curve is essentially tangent for 16.5 feet, with an angle of just less than 2 degrees to the main track. This geometry results in relatively high lateral forces, particularly when the train traverses the diverging route through the switch. For comparison, the AREA recommends a transition spiral length of 93 feet for a 12-degree curve with a revenue service speed of 19 mph. The spiral geometry allows a less sudden transition for the wheelsets.

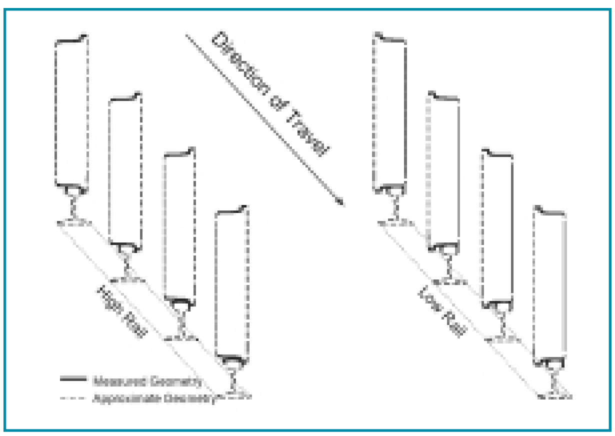




The figure shows an illustration of the wheel/rail contact geometry, with the rail and the switch point in both the new and worn conditions. As the gage faces of the switch point and rail wear, the contact point moves further down the flange of the wheel. As a result, for the worn condition, the top of the wheel flange contacts the switch point and the side of the rail. When there is significant wear of the gage faces of the switch point and rail, the wheel/rail contact angle is the angle of the gage faces of the switch point and the side of the rail.

KEY FINDINGS

Preliminary results of geometric analysis of the wheel and rail geometry indicate that changes in rail head profile may allow the wheel to climb the rail.



More detailed analyses are currently being evaluated, including dynamic effects and wheel/rail interaction. Means of extending current models to allow variation of the rail head profile as a function of distance are being investigated.

FUTURE RESEARCH

Currently, FRA efforts in vehicle/track interaction are primarily directed toward implementing the track surface geometry algorithms in a real-time track surface geometry measurement and evaluation system, experimentally determining the safe limits of track twist, and determining the safe envelope for track gage and alignment geometry. Developing this envelope requires an understanding of how alignment variations can influence rail vehicle response to crosslevel variations. Analytic models are being constructed, exercised, and compared with existing track alignment and gage specifications. The research to be conducted in the next 5 years is expected to include a determination of the safe envelope for gage and track alignment geometry, and a determination of the influence of gage and track alignment variations on vehicle response to profile and crosslevel variations.

HIGH-SPEED TRACK GEOMETRY SPECIFICATIONS

New high-speed equipment standards that cover equipment operating speeds from 110 mph to 200 mph have been developed by the FRA. The incorporation of high-speed operation introduces new requirements on vehicles, geometry standards for gage, surface, and alignment, and the track structure to minimize the potential for unsafe operating conditions. While some standards are identical to their counterparts in lower track classes, several sections are unique to the high-speed environment.

When an encounter with a track geometry variation or series of variations happens at high speed, an unsafe vehicle response such as excessive carbody acceleration or derailment can occur. Large surface variations in track geometry can cause carbody pitch and bounce, resulting in unsafe carbody accelerations or wheel unloading. Track alignment and gage variations can lead to large lateral wheel and axle forces, resulting in derailment or damage to the track structure.

RESEARCH STATUS

Research has been conducted to identify combinations of surface alignment and gage amplitude and wavelength irregularities that cause excessive accelerations in a vehicle carbody or wheel/rail forces. Locomotive designs have been examined since they could present the largest problem because of their weight. For high-speed operation, locomotives can be designed with traction motors mounted to the carbody or to the truck frame. Both of these potential designs were examined.

A computer model has been developed to determine the minimum amplitude of track surface variation required to cause excessive vertical accelerations (0.6 g) in the locomotive operator's cab. The model has four degrees of freedom (carbody pitch and bounce, and vertical displacements of the front and rear trucks). This model was used to examine locomotives with suspension characteristics and inertial properties representative of those for potential use at high speed. The influence of speed on vehicle response to isolated and repeated track surface variations was determined for a wide range of wavelengths. The influences of equipment suspension parameters, such as secondary suspension damping, were also determined.

The NUCARS simulation program was used for the analyses of equipment response to track alignment variations. The amplitude of a single perturbation required to cause excessive lateral carbody accelerations, wheelclimb and large wheel rail lateral over vertical forces was determined for a range of wavelengths.

KEY FINDINGS

The surface analysis results indicate that equipment suspension parameters and configuration strongly influence vehicle response to track geometry variations. In particular, mounting of the traction motors strongly influences vehicle response to track geometry, especially at speeds greater then 125 mph. The analysis results showed that a locomotive design with truck-mounted traction motors requires approximately 33 percent smaller track profile variation amplitude to cause excessive vertical accelerations in the operator's cab than a locomotive design with carbody-mounted traction motors. The results indicate that a locomotive design with truck-mounted traction motors will exceed 0.6 g peak-to-peak acceleration in the operator's cab for isolated 1 inch track profile geometry variations at a speed of 160 mph. These isolated variations range in wavelength from 30 to 100 feet.

The alignment studies indicate that at short wavelengths (less than about 100 feet), the maximum safe amplitude of alignment variation is limited by the wheel rail lateral over vertical forces. At long wavelengths (those greater than 100 feet), safe amplitude of alignment variation is limited by carbody accelerations.

These studies have been used to support the development of high-speed track geometry standards.

FUTURE RESEARCH

Work in the area of High-Speed Track Geometry Specifications will continue in three specific areas. First is the continued mapping of vehicle response to geometry variations and combinations of variations. Also important in this area is the treatment of innovative truck, car, and trainset designs, including passive, frequency tuned and fully active steering and suspension elements and articulated vehicle connections. Second is the detailed investigation of the influence of specific wheel and rail profiles and the importance of controlling the contact geometry, especially in proximity to special track features such as turnouts. Finally, studies will be conducted to determine how best to focus inspection technologies to identify incipient track geometry conditions before reaching safety critical amplitudes. An important aspect of this research is to develop a comprehensive strategy for assuring adequate geometry.